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Water cooled minichannel heat sinks for microprocessor cooling: Effect of fin spacing

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HIGHLIGHTS

- Five heat sink geometries were tested to investigate the effect of fin spacing on microprocessors' operating temperature.
- Microprocessor heat was simulated using a heating block with 325 W power.
- Decreasing the fin spacing of the heat sinks increased overall heat transfer coefficient while thermal resistance decreased.
- 0.2 mm fin spacing heat sink produced the lowest base temperature of 40.5 °C at a heater power of 325 W.

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ABSTRACT

For effective thermal management of high heat generating microprocessors, five different heat sinks with fin spacings of 0.2 mm, 0.5 mm, 1.0 mm, and 1.5 mm along with a flat plate heat sink were investigated. Microprocessor heat was simulated by a heated copper block with water as a coolant. At a heater power of 325 W, the lowest heat sink base temperature of 40.5 °C was achieved by using a heat sink of 0.2 mm fin spacing which was about 9% lower than the best reported base temperature of 44 °C using a nanofluid with commercial heat sink in the open literature. The base temperature and thermal resistance of the heat sinks were found to drop by decreasing the fin spacing and by increasing volumetric flow rate of water circulating through the heat sink. For a flat plate heat sink, the maximum thermal resistance was 0.216 K/W that was reduced to as little as 0.03 K/W by using a heat sink of 0.2 mm fin spacing. The overall heat transfer coefficient was found to be 1297 W/m^2 K and 2156 W/m^2 K for the case of a flat plat and 0.2 mm fin spacing heat sinks, respectively, the latter showed about two-folds enhancement compared to the former.

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1. Introduction

For the effective thermal management of high heat generating computer processors and to maintain operating temperature in a range of 60–80 °C, new and novel cooling techniques are being used. Air cooling techniques have reached their limit in heat removing capabilities. In recent years, attention is now focused on liquid cooling techniques due to higher heat transfer coefficients associated with liquids. Generally two approaches have been employed in order to optimize the performance of liquid cooling systems. The first approach was to modify the heat sink geometry using ordinary coolants while the second approach involved the modification of thermophysical properties of ordinary fluids in an effort to enhance their heat transfer performance.

A numerical study of the heat transfer characteristics of water cooled minichannel heat sinks was performed by Xie et al. [1] and concluded that the heat removed by a heat sink increases by decreasing the channel width of minichannels while the thermal resistance increases by increasing the channel width. Whelan et al. [2] designed a liquid cooling system for CPU cooling by considering the cost and ease of manufacture. This new block based on miniature jet stream design performed better than the commercially available cooling block [2]. In an effort to optimize the commercial liquid cooling systems, Naphon and Wongwises [3] performed an experimental analysis on the jet liquid heat transfer characteristics of the mini rectangular fin heat sink of a CPU based on real processor operating conditions and they concluded that the use of jet impingement cooling system resulted in a lower CPU operating

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temperature in comparison with conventional liquid cooling system. In order to increase the efficiency of liquid cooling systems further, an intermittent multi jet spray system for the cooling of microprocessors was employed by Panão et al. [4] which resulted in intelligent thermal management. A control strategy was devised based on the Intermittent Spray Cooling concept [4]. Bower et al. [5] experimentally investigated the heat transfer in water cooled silicon carbide milli-channel heat sink and they concluded that liquid cooled SiC heat sinks outperform air cooled heat sinks and compared favorably with copper equivalents, therefore a suitable option for better thermal management of electronics. Numerical techniques have also been employed to investigate the heat transfer characteristics of cooling systems for CPUs. Naphon et al. [6] conducted a numerical investigation to evaluate the heat sink's cooling performance based on real PC operating conditions.

Another aspect of enhanced liquid cooling that has received a great attention in recent years is the enhancement of thermophysical properties of liquids. This is mainly achieved by suspending nano meter sized particles in ordinary fluids and forming a stable suspension to produce nanofluids. Eastman et al. [7] reported a 40% enhancement in thermal conductivity as a result of 0.3% volume loading of copper nanoparticles in ethylene glycol. Choi et al. [8] also reported an anomalous enhancement of 160% in thermal conductivity at 1% volume concentration of Multi Walled Carbon Nanotube (MWCNTs) in oil. The work of Xie et al. [1] was extended by Ijam and Saidur [9] in which they compared the performance of water cooled and nanofluid cooled copper minichannel heat sinks and concluded that nanofluids give better heat transfer performance.

The use of nanofluids for computer cooling applications has also been investigated. For this purpose commercially available liquid cooling kits have been employed. Some of the investigations involve mounting the cooling block over the actual processor while in other investigations the cooling blocks were attached to a heating block that simulated the processor heat. The later approach shows more control on the experimental conditions. The performance of Al_2O_3–water nanofluid in commercial liquid cooling system was evaluated by Roberts and Walker [10] and reported an enhancement of approximately 20%. They also observed that the temperature gained by nanofluids while passing through the cooling blocks was greater as compared to water passing through the same block at the same processor heat flux. Nguyen et al. [11] using Al_2O_3 nanofluid in jet type cooling block reported an enhancement of 38% in convective heat transfer coefficient. Nguyen et al. [12] reported for turbulent conditions, there was a greater decrease in the junction temperature as compared to the laminar flow regime when ethylene glycol based Al_2O_3 nanofluids were used. The use of oscillating heat pipe and nanofluids has been investigated for cooling of electronics. The use of diamond nanofluids with 1% volume loading was investigated by Ma et al. [13] and as a result they were able to reduce the temperature difference between the evaporator and the condenser from 40.9 to 24.3 °C at an input power of 80.0 W.

Rafati et al. [14] recently has reported an experimental investigation using a commercial liquid cooling kit (3D Galaxy II by Gigabyte). They used a base fluid made of 75% water and 25% of ethylene glycol. TiO_2, SiO_2 and Al_2O_3 nanoparticles with different volume loading in the base fluid was tested to achieve the lowest operating temperature of a quad-core processor (Phenom II X4 965 with thermal design power 125 W). Pure base fluid produced a temperature drop from 53.3 °C to 49.3 °C for flow rates of 0.5 L per minute to 1.0 L per minute, respectively. The lowest temperature drop was found using Al_2O_3 nanofluids with a 1% volume concentration in base fluid, which was 46.1–43.9 °C for flow rates of 0.5 L per minute to 1.0 L per minute, respectively.

From the above literature review it is revealed that researchers have used either commercial minichannels with pure liquids or nanofluids with commercial cooling kits to reduce the microprocessor temperature. No attempt has been made to investigate the systematic effect of sink geometry on the microprocessor temperature. On the other hand, the use of nanofluids is still ambiguous due to many problems associated with them, i.e. more maintenance required, higher cost, aggregation and deposition of nano particles [15]. In the present study, an attempt has been made to investigate the systematic effect of sink geometry (by reducing fin spacing) with water as fluid on the microprocessor base temperature. A cylindrical copper block with high power of 325 W is used to simulate a high heat generating microprocessor. The objective of the present study was to identify the potential of heat sink geometries that are sufficient to reduce the high heat generating (up to 325 W) microprocessors temperature to a safe operating value in comparison of the commercially available heat sinks and nanofluids.

2. Experimental apparatus

2.1. Heat sinks

To determine the suitable heat sink type for the thermal management of high heat generating microprocessors, five different copper heat sink geometries were tested. Four heat sinks (see Fig. 1) had a varying fin spacing of 0.2 mm, 0.5 mm, 1.0 mm and 1.5 mm while the fifth heat sink had a flat surface for comparison. The finned heat sinks were manufactured by electrode discharge machining (EDM) while the flat plate was manufactured by milling operation. The fin thickness and height was kept constant at 1 mm and 3 mm, respectively. The base of all the heat sinks had a square protrusion of 0.5 mm and dimensions of 28.7 mm × 28.7 mm. This protruded square base was used to mount the heat sinks on the heating block that simulated the microprocessor heat. Table 1 shows the dimensions of the heat sinks. The copper heat sinks were sealed using plexi-glass and rubber seals in such a way that fins tips were in complete contact to the plexi-glass allowing no fluid flow over the fin tips. Inlet and outlet nozzles were provided in the final heat sink assembly for the coolant to flow through the heat sinks.

2.2. Test loop

To simulate the processor, a heating block assembly was used as shown in Fig. 2. The reason for using the heating block was that processor heat varies with varying processor loadings and in this study a constant heat of 325 W was applied to copper block by two surface heaters. The heaters clamped the cylindrical copper block from the outside. In order to ensure constant power being delivered to the heater, a DC power supply (Loadstar 8109) was used which supplied constant voltage of 197 V and current of 1.65 A to give a total power of 325 W. The heat sinks were mounted on the top of the heated copper block and a heat sink thermal compound was applied between the two surfaces in order to ensure a perfect thermal contact. All exposed surfaces were insulated by using fiberglass wool.

A commercial CPU liquid cooling system (Galaxy by Gigabyte) was used in the present investigation. Fig. 3 shows the schematic diagram of the experimental setup. The test loop was modified to include a needle valve in order to control the flow of water and three different volumetric flow rates of 0.5 LPM, 0.75 LPM and 1.0 LPM were employed. The flow rate was measured by using an Omega rotameter (full scale accuracy: ±5%).

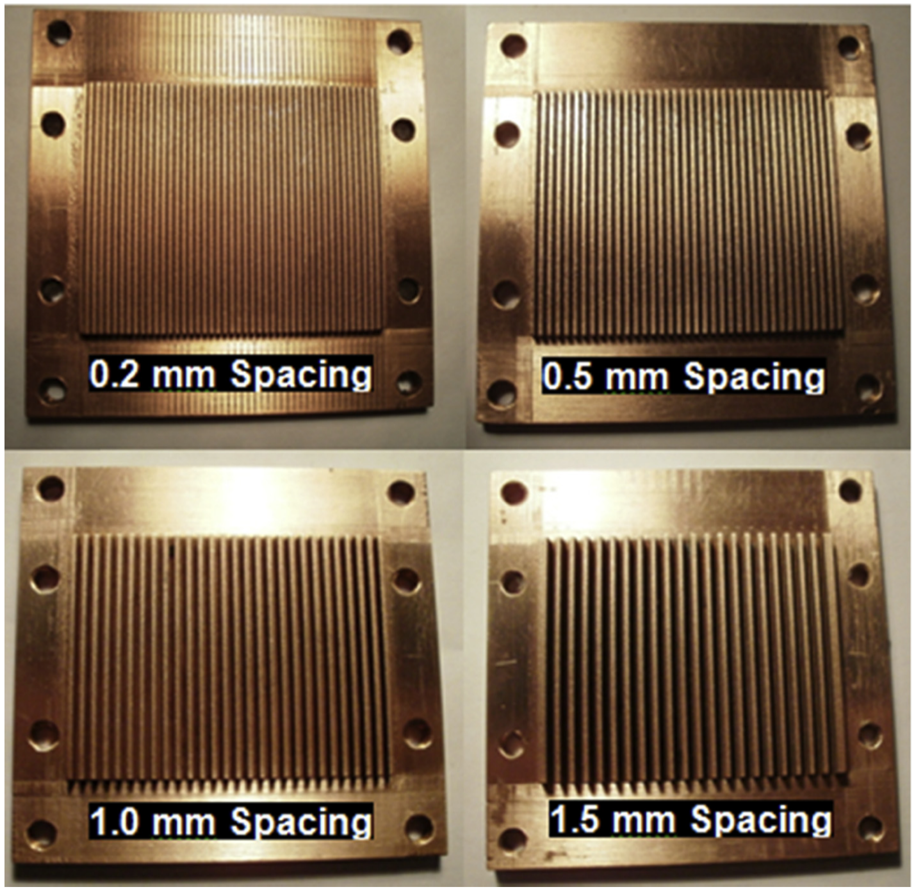

Fig. 1. Heat sink geometries.

In order to measure T_{in}, T_{out} and T_{base}, three K-type thermocouples (Omega TC-TT-KI-24-1M) were used. These thermocouples were calibrated in a constant temperature water bath against a PT-100 temperature sensor and they were accurate to within ± 0.1 K. Steady state values of T_{in}, T_{out} and T_{base} were used in calculations. The steady state was identified when the temperature measurements changed by no more than ± 0.1 °C in 5 min.

2.3. Data reduction

The heat removed by the water circulating through the heat sink is calculated by applying the conservation of energy principle to control volume with one inlet and one exit, with no work interactions and assuming negligible changes in kinetic and potential energies. This is expressed by Eq. (1)

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \tag{1}$$

The inlet and outlet water temperatures are measured by two K-type thermocouple probes inserted into the insulated pipes connected to the heat sink. The thermophysical properties of water at mean fluid temperature, $T_{m,f}$, Eq. (2) are used in all of the calculations.

$$T_{m,f} = \frac{(T_{in} + T_{out})}{2} \tag{2}$$

The log-mean temperature difference (LMTD) across the heat sink was calculated by Eq. (3). T_{base} was measured by a fine wire K-type thermocouple inserted at the junction of the heated copper block and copper heat sink.

$$\mathrm{LMTD} = \frac{(T_{base} - T_{in}) - (T_{base} - T_{out})}{\ln\left(\frac{(T_{base} - T_{in})}{(T_{base} - T_{out})}\right)} \tag{3}$$

The overall heat transfer coefficient for each heat sink for every flow rate was calculated by applying Eq. (4) which reduces to give Eq. (5).

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = UA(\mathrm{LMTD}) \tag{4}$$

$$U = \frac{\dot{m}c_p(T_{out} - T_{in})}{A(LMTD)} \tag{5}$$

The heat transfer area of the finned heat sink was calculated by Eq. (6) based on the dimensions as shown in Fig. 4.

$$A_{fin} = 2(A \cdot B) + (c \cdot l)(n-1) + (h \cdot l \cdot 2)n + (t \cdot h \cdot 2)n \tag{6}$$

Similarly to calculate the heat transfer area for the flat plate heat sink, Eq. (7) was used.

Table 1
Dimension of heat sinks (mm).

Length of fins (l)	45
Width of finned section on the heat sink (A)	55
Unfinned length of heat sink (B)	10
Height of fins (h)	03
Thickness of fins (t)	01
Thickness of heat sink base plate	03

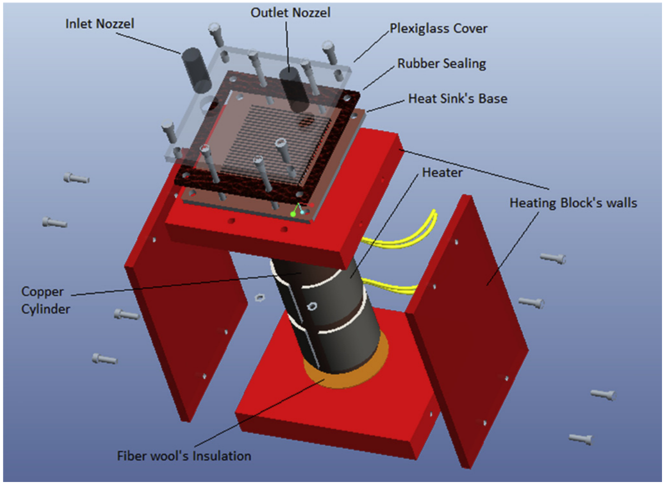

Fig. 2. Schematic of heating block assembly.

$$A_{flat} = A \cdot (l + 2B) \tag{7}$$

Table 2 shows the active area for heat transfer of all five heat sinks.

The thermal resistance for heat sinks was computed according to Eq. (8) [2].

$$R_{th} = \frac{LMTD}{\dot{Q}} \tag{8}$$

For calculating the enhancement in area in comparison to flat plate heat sink by using finned geometries Eq. (9) is used.

$$A_{en} = \frac{\text{Active Area of Finned heat sink}}{\text{Active Area of flat plate}} \tag{9}$$

Similarly, to compute the enhancement in overall heat transfer coefficient in comparison to flat plate heat sink as a result of using finned heat sinks Eq. (10) is used.

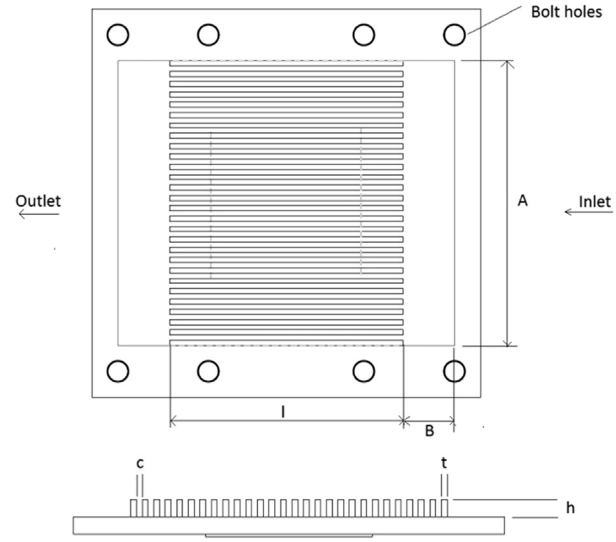

Fig. 4. Schematic of heat sink geometry.

$$U_{en} = \frac{\text{Overall heat transfer coefficient for finned geometry}}{\text{Overall heat transfer coefficient for flat plate heat sink}} \tag{10}$$

2.4. Uncertainty analysis

A method developed by Kline and McClintock [16] was used here to estimate the uncertainties in experimental data. This method incorporates the estimated uncertainties in the experimental measurements (e.g. coolant flow rate, coolant inlet and outlet temperatures and heat sink base temperature) into the final parameters of interests (e.g. heat transfer rate and overall heat transfer coefficient, etc.). Using the measured uncertainties of the above-mentioned parameters, the maximum calculated

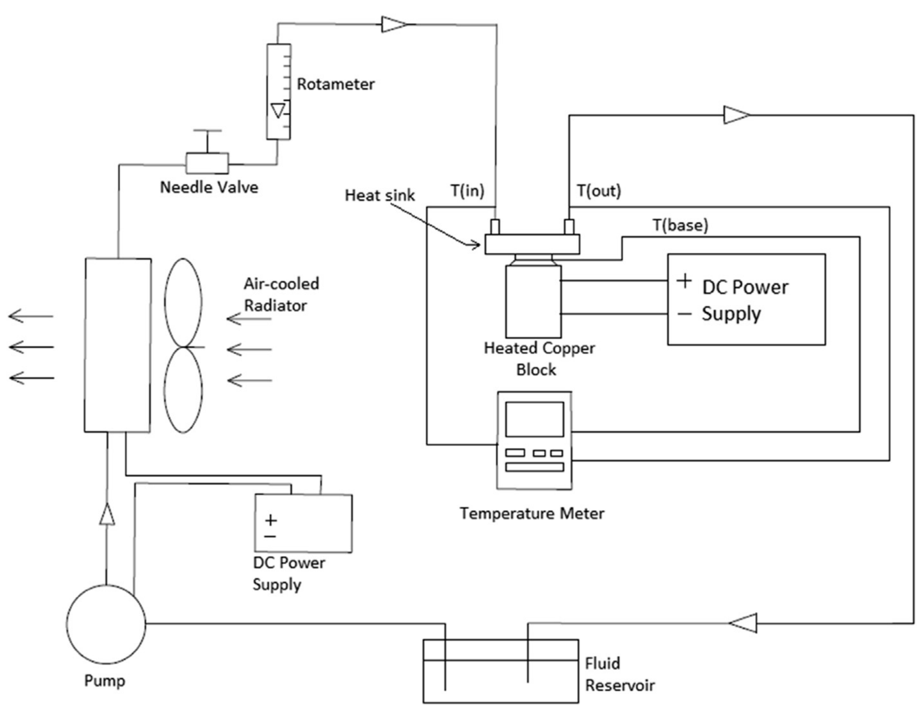

Fig. 3. Schematic of experimental setup.

Table 2
Active Area for heat transfer of heat sinks.

Fin spacing (mm)	Active area (mm^2)
0.2	13,931
0.5	11,852
1.0	9773
1.5	8319
Flat plate	3575

uncertainties in heat transfer rate were found to be 4.0% and 3.9% for plain and finned heat sinks, respectively. The maximum uncertainty in overall heat transfer coefficient was about 5.8% and 4.7% for plain and finned heat sinks, respectively, the coolant outlet temperature participated significantly in calculated uncertainty.

3. Results

3.1. Heat sink geometry and base temperature

The base temperature of the heat sink is analogous to the processor operating temperature under varying processor-loading conditions. Fig. 5 shows variation of base temperature with the flow rate of water circulating through the five heat sink geometries. For flat plate heat sink, the base temperature value is as high as 91.3 °C at a flow rate of 0.5 LPM. For all five sinks, the base temperature drops by increasing the flow rate of the coolant through the heat sink. The base temperature drops considerably in comparison to flat plate when finned geometries are used. It can be seen from Fig. 5 that the effect of coolant flow rate on base temperature is much more dominant for the case of a flat plate compared to the finned heat sinks (which is due to a higher increase in Reynolds number with flow rates in comparison finned sinks, see Fig. 6). Decreasing fin spacing considerably reduces the effect of flow rate on base temperature (lower Reynolds number). For example, changing the flow rate from 0.5 LPM to 1.0 LPM produces a drop in base temperature as 14.6 °C and 3 °C for the case of flat plate and 0.2 mm fin spacing heat sink, respectively. Fig. 5 also compares the present data with the data reported by Rafati et al. [14] for a base fluid of water and ethylene glycol and 1% Al_2O_3 volume concentration nanofluid in base fluid used with a commercial heat sink. It can be seen that present heat sink geometry with a fin spacing of 1.0 mm was giving about same base temperatures as reported by Rafati et al. [14] for a base fluid of ethylene glycol and water used with a commercial heat sink. Also, a present geometry with a fin spacing of 0.2 mm showed even superior performance (about 10% reduction in base temperature) as compared to the base temperatures achieved by using 1% Al_2O_3 volume concentration nanofluid at the same flow rates reported by Rafati et al. [14]. It should be noted here that present study was performed by simulating a microprocessor with a heat generation of 325 W whereas the study reported by Rafati et al. [14] was performed with a microprocessor with thermal design power of 125 W. It is clear from this comparison that geometrically enhanced heat sinks still have a great potential to be used with ordinary fluid like water to reduce the high heat generating microprocessors temperature to a safe value.

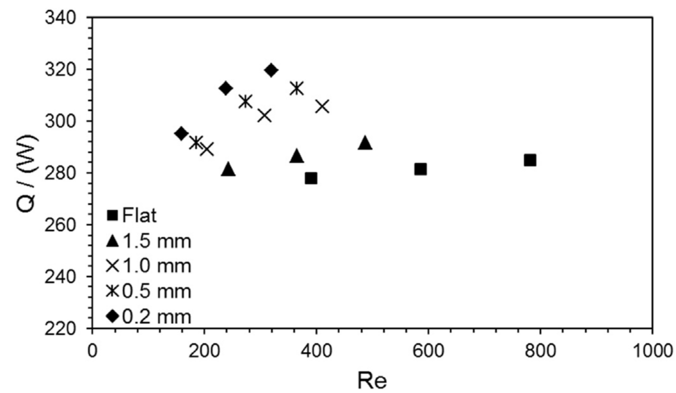

Fig. 6. Comparison of heat transfer rate with Reynolds number as a function of fin spacing.

Fig. 6 shows a comparison of heat transfer rate to the coolant with Reynolds number as a function of fin spacing. The highest range (400–800) of Reynolds number was found for flat heat sink which decreased significantly with decreasing the fin spacing (within 180–500 for all finned sinks). Despite the smaller values of Reynolds number for the finned sinks, they showed significantly higher heat transfer rates in comparison of flat heat sink which was due to the fact of significant area enhancement (from 2.3 to 4 in comparison of flat heat sink for 1.5–0.2 mm finned heat sinks, respectively, see Fig. 9). It can be seen that a finned heat sink with fin spacing of 0.2 mm at a Reynolds number of about 320 (flow rate of 1 L per minute) was able to remove 320 W of heat (about 98% of the total supplied power which was 325 W). The minimum heat removed found by the flat sink at a flow rate of 0.5 L per minute was 277 W (about 85% of the supplied power). A small portion of heat loss from heating block to the outside air through insulation was calculated assuming one-dimensional radial heat flow. The total supplied power was always found to be within 2% error of heat absorbed by the coolant plus heat loss to the outside air from heating block through insulation.

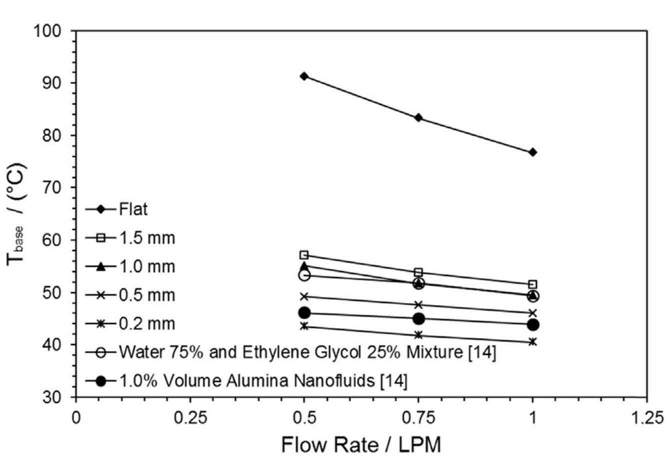

Fig. 5. Variation of base temperature of heat sinks with volumetric flow rate.

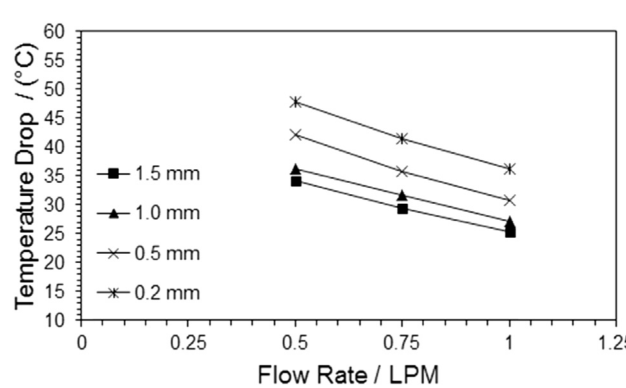

Fig. 7. Comparison of temperature drop with volumetric flow rate.

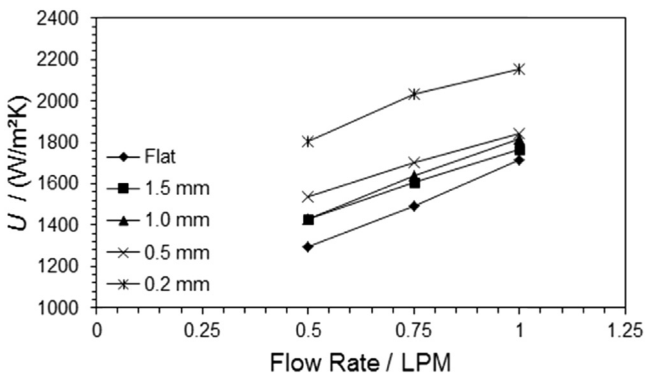

Fig. 8. Variation of overall heat transfer coefficient with volumetric flow rate.

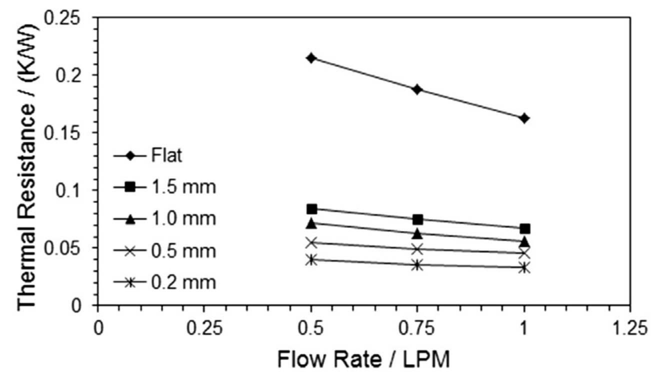

Fig. 10. Variation of thermal resistance of heat sinks with volumetric flow rate.

Fig. 7 shows the drop in base temperature produced by finned geometries in comparison to the flat plate heat sink. The greatest base temperature drop of 47.8 °C at 0.5 LPM resulted when 0.2 mm fin spacing heat sink was used. As a result of using minichannel geometries of varying fin spacing, the lowest base temperature of 40.5 °C was achieved at a flow rate of 1 LPM by using the 0.2 mm fin spacing heat sink. Achieving a base temperature as low as 40.5 °C when the processor is generating heat in excess of 300 W ensures safe and reliable operation of the high performance processor.

3.2. Heat sink geometry and overall heat transfer coefficient

Fig. 8 shows the overall heat transfer coefficient with variation in volumetric flow rate of water for all five heat sinks. The maximum value of overall heat transfer coefficient was found as 2156 $W/m^2 K$ at volume flow rate of 1 LPM corresponding to the heat sink with fin spacing of 0.2 mm. Meanwhile the lowest value of overall heat transfer coefficient of 1297 $W/m^2 K$ at a volume flow rate of 0.5 LPM was observed for the case of flat plate heat sink. It can be seen that overall heat transfer coefficient increases with decreasing fin spacing. The trend line of overall heat transfer coefficient for 0.2 mm fin spacing heat sink is considerably higher than the other geometries. This is due to the fact of largest heat transfer area for this particular geometry heat sink (see Table 2). Fig. 9 shows variation of heat transfer area enhancement and the average enhancement in overall heat transfer coefficient with respect to the flat plate heat sink as a consequence of finned heat sinks. The greatest enhancement of 1.39 is observed in overall heat transfer coefficient which corresponds to an area enhancement of 3.9 as a result of using the heat sink with fin spacing of 0.2 mm.

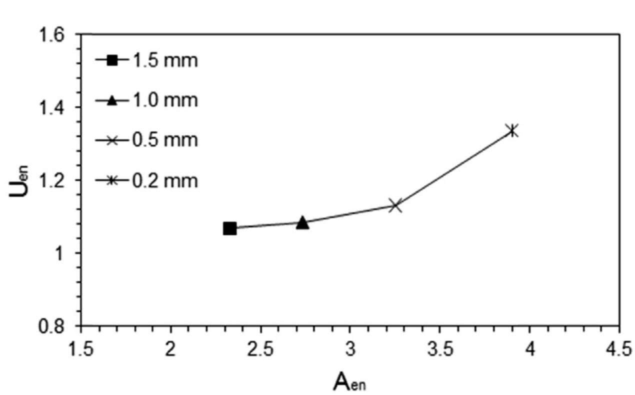

Fig. 9. Average overall heat transfer coefficient enhancement with active area enhancement.

3.3. Thermal resistance of heat sinks

Fig. 10 shows the variation in thermal resistance with volume flow rate for all five heat sink geometries. The thermal resistance for all heat sinks decreases by increasing the volumetric flow rate of water. The values of thermal resistance for flat plate heat sink geometry are considerably higher in comparison to the finned heat sink geometries. The maximum value of thermal resistance was found to be 0.216 K/W for flat plate heat sink at 0.5 LPM. The thermal resistance decreases by decreasing the fin spacing for any given volumetric flow rate of the water. Heat sink with a fin spacing of 0.2 mm produced the lowest value of thermal resistance of 0.033 K/W at a volume flow rate of 1 LPM.

4. Conclusion

Five different heat sinks with fin spacings of 0.2 mm, 0.5 mm, 1.0 mm, and 1.5 mm along with a flat plate heat sink were investigated for effective thermal management of high heat generating microprocessors. At a simulated microprocessor power of 325 W, the lowest heat sink base temperature of 40.5 °C was achieved by using a heat sink of 0.2 mm fin spacing at a flow rate of 1.0 LPM which was 9% lower than the best reported value of base temperature of 40 °C (Phenom II X4 965 with thermal design power of 125 W) by using a nanofluid with a commercially available heat sink [14]. It was also found that a finned heat sink with 1.0 mm of fin spacing was capable to produce the same base temperature as was achieved by a commercially available heat sink using a mixture of water and ethylene glycol [14]. It can be concluded from this experimental investigation that geometrically enhanced heat sinks by using with freely available fluid, i.e. water still have a lot of potential to cool the high heat generating microprocessors (more than 300 W). Manufacturers should focus their attention to fully exploit the heat transfer capabilities of enhanced geometries used with water despite using the nanofluids whose performance is still ambiguous due to many problems (e.g. agglomeration, deposition, more maintenance, high cost, etc.) associated with them.

Nomenclature

A	width of finned section of heat sink [mm]
A_{en}	area enhancement
A_{fin}	heat transfer area of the finned heat sink [m^2]
A_{flat}	heat transfer area of the flat plate heat sink [m^2]
B	unfinned length of the heat sink [mm]
c	fin spacing [mm]
c_p	specific heat [kJ/kg °C)
h	height of fins [mm]
LMTD	log-mean temperature difference

LPM liter per minute
l length of fins [mm]
$\dot{m}$ mass flow rate of water circulating through the heat sink [kg/s]
n number of fins
$\dot{Q}$ heat removed by the water circulating through the heat sink [W]
R_{th} thermal resistance of heat sink [K/W]
T_{base} base temperature of heat sink [°C]
T_{in} temperature of water at heat sink inlet [°C]
$T_{m,f}$ mean fluid temperature [°C]
T_{out} temperature of water at heat sink outlet [°C]
t fin thickness [mm]
U overall heat transfer coefficient [W/m^2 K]
U_{en} overall heat transfer coefficient enhancement

Subscripts
en enhancement
f fluid
fin finned heat sink
flat flat plate heat sink
m mean
th thermal

References

[1] X.L. Xie, W.Q. Tao, Y.L. He, Numerical study of turbulent heat transfer and pressure drop characteristics in a water cooled minichannel heat sink, J. Electron. Packag. 129 (2007) 247–255.
[2] B.P. Whelan, R. Kempers, A.J. Robinson, A liquid-based system for CPU cooling Implementing a jet array impingement waterblock and a tube array remote heat exchanger, Appl. Therm. Eng. 39 (2012) 86–94.
[3] P. Naphon, S. Wongwises, Investigation on the jet liquid impingement heat transfer for the central processing unit of personal computers, Int. J. Heat Mass Transfer 37 (2010) 822–826.
[4] M.R.O. Panão, J.P.P.V. Guerreiro, A.L.N. Moreieira, Microprocessor cooling based on an intermittent multijet spray system, Int. J. Heat Mass Transfer 55 (2012) 2854–2863.
[5] C. Bower, A. Ortega, P. Skandakumaran, R. Vaidyanathan, C. Green, T. Philips, Heat transfer in water- cooled silicon carbide milli-channel heat sinks for high power electronic applications, in: Proceedings of IMECE'03: 2003 ASME International Mechanical Engineering Congress and Exposition, Washington, D.C., November 16–21, 2003.
[6] P. Naphon, S. Klangchart, S. Wongwises, Numerical investigation on the heat transfer and flow in the mini-fin hat sink for CPU, Int. Commun. Heat Mass Transfer 36 (2009) 834–840.
[7] J.A. Eastman, S.U.S. Choi, S. Li, W. Yu, L.J. Thompson, Anomalously increased effective thermal conductivities of ethylene glycol based nanofluids containing copper nanoparticles, Appl. Phys. Lett. 78 (2001) 718, http://dx.doi.org/10.1063/1.1341218.
[8] S.U.S. Choi, Z.G. Zhang, W. Yu, F.E. Lockwood, E.A. Grulke, Anomalous thermal conductivity enhancement in nanotube suspensions, Appl. Phys. Lett. 79 (2001) 2252, http://dx.doi.org/10.1063/1.1408272.
[9] A. Ijam, R. Saidur, Nanofluid as a coolant for electronic devices (cooling of electronic devices), Appl. Therm. Eng. 32 (2012) 76–82.
[10] N.A. Roberts, D.G. Walker, Convective performance of nanofluids in commercial electronics cooling applications, Appl. Therm. Eng. 30 (2010) 2499–2504.
[11] C.T. Nguyen, G. Roy, C. Gauthier, N. Galanis, Heat transfer enhancement using Al_2O_3 –water nanofluid for an electronic liquid cooling system, Appl. Therm. Eng. 27 (2007) 1501–1506.
[12] C.T. Nguyen, G. Roy, P.R. Lajoie, S.E.B. Maiga, Nanofluids heat transfer performance for cooling of high heat output microprocessor, in: Proceedings of the Third IASME/WSEAS International Conference on Heat Transfer, Thermal Engineering and Environment, Greece, 2005, pp. 160–165.
[13] H.B. Ma, C. Wilson, Borgmeyer, K. Park, Q. Yu, et al., Effect of nanofluid on the heat transport capability in an oscillating heat pipe, Appl. Phys. Lett. 88 (2006) 143116, http://dx.doi.org/10.1063/1.2192971.
[14] M. Rafati, A.A. Hamidi, M.S. Niaser, Applications of nanofluids in computer cooling systems (heat transfer performance of nanofluids), Appl. Therm. Eng. 45–46 (2012) 9–14.
[15] L. Godson, B. Raja, D. Mohan Lal, S. Wongwises, Enhancement of heat transfer using nanofluids – an overview, Renew. Sustain. Energy Reviews 14 (2010) 629–641.
[16] S.J. Kline, F.A. McClintock, Describing uncertainties in single-sample experiments, Mech. Eng. 75 (1953) 3–8.